PUMPABILITY OF AIRCRAFT TURBINE LUBRICANTS AT LOW TEMPERATURES

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by E. S. Starkman* & J. H. Bridges**

Questions regarding the principles governing flow characteristics of lubricating oils in the vicinity of the pour point have led to an investigation into the behavior of a series of typical aircraft gas turbine lubricants in a simple system at temperatures down to −70 F. The lubricants tested included two oils of military specification MIL-O-6081B (a 1005 grade and a 1010 grade), a sample equivalent in other properties to 1005 grade but having a −15 F. pour point, di(2-ethylhexyl) sebacate, and an SAE 10W motor oil with a −30 F. pour point. Results indicated that laminar flow relationships were satisfactory for predicting flow characteristics down to the pour point so long as proper attention was paid to pump inlet conditions. In addition, there was evidence that fluids with so-called waxy pour points could be satisfactorily pumped to temperatures lower than the pour point, a conclusion which may be of significance.

Nomenclature

d	=	diameter, feet
x	=	distance in direction of flow, feet
L	=	length of conduit, feet
μ	=	coefficient of viscosity, pound seconds per square foot
p	=	pressure, pounds per square foot
Δp	=	pressure drop, pounds per square foot
r	=	radius, feet
R	=	hydraulic radius, feet
τ	=	shear stress, pounds per square foot
v	=	local velocity, feet per second
v_m	=	maximum velocity, feet per second
V	=	average velocity, feet per second
y	=	distance normal to flow, feet

The basic objective of this investigation was to determine, insofar as possible, the flow characteristics for representative gas turbine lubricants in a simple hydrodynamic system in the vicinity of their pour points. The need for such an investigation was made apparent by previous attempts to evaluate[1] and design[2] engine systems, and by an academic interest in obtaining information on the behavior of certain specific hydrocarbon and non-hydrocarbon lubricants at temperatures near their so-called pour point.[3] Some limited information was available in the literature,[4, 5] but this was not felt sufficiently representative of the problem of pumping low viscosity lubricants in aircraft plumbing.

The successful design of any flow system, whether for application to a single turbojet engine, for an integral oil transfer system for multi-engine aircraft,[6] or even for ground handling,[7] depends upon some knowledge of the quantity of energy necessary to move the fluid under the most adverse conditions. This, in turn, depends upon the fundamental laws governing flow, and particularly on the shear stress relationships in the fluid.

The relationship between shear stress and fluid flow for certain systems has been well established for more than 100 years.[8] The specific goal of this investigation was to ascertain experimentally if these relationships were applicable to typical turbine lubricants in a simple but representative flow system under low temperature conditions.

The problem of providing a bearing with satisfactory lubricant at relatively high temperatures and, also, insuring that the lubricant has fluidity at low temperatures, is one which is faced in many engineering applications. In the case of aircraft turbine engines, the problem is specifically that of adequately lubricating the hot turbine bearing with a fluid which must also be pumpable at the low temperatures encountered in Arctic or high altitude operation. The character of the oil for a particular application is fixed by the load requirement at high temperature and, because of the first consideration being that of lubrication, the design of pumps and plumbing becomes completely dependent on the resulting low temperature properties of the lubricant chosen.

In providing an adequate supply system for low temperatures, and particularly for temperatures in the vicinity of the pour point of a lubricant, the question arises as to proper design criteria. A resolution of the problem depends upon whether or not the lubricant acts as a Newtonian fluid, and/or follows laminar flow relationships. Algebraic definition of such conditions is familiar, but will be repeated here to indicate the specific criteria used in making conclusions later.

A Newtonian fluid is one which, by definition,[9] undergoes movement such that the product of its viscosity and velocity gradient is equal to its shear stress anywhere in the flow field. It follows directly, therefore, that Newtonian flow may be expressed as

$$\tau = \mu \frac{dv}{dy} \qquad (1)$$

If the system in which the flow occurs is a horizontal cylindrical flow channel, the pressure gradient may also be simply related to the shear

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stress as follows

$$\frac{dp}{dx} = \frac{\tau}{R} \tag{2}$$

which, when integrated for a flow path of length L, assuming τ and R to be constant, and recognizing the hydraulic radius R to be $r/2$, becomes

$$\tau = \left(\frac{\Delta p}{L}\right)\frac{r}{2} \tag{3}$$

Further, assuming velocity and shear stress distribution to be cosymmetric to the pipe, Eqs. 1 & 3 may be related to give

$$\tau = \mu\frac{dv}{dy} = \left(\frac{\Delta p}{L}\right)\frac{r}{2} \tag{4}$$

and separating variables yields

$$dv = \frac{\Delta p}{\mu L}\frac{r}{2}dy \tag{5}$$

The velocity distribution for a cylindrical channel can be obtained from Eq. 5 by taking y decreasing and r increasing from the centerline of the cylinder, allowing $v = v_m$ at $r = 0$, and integrating to give

$$v = v_m - \frac{\Delta p}{4\mu L}r^2 \tag{6}$$

Since it can be shown[10] that the velocity distribution in Eq. 6 is parabolic, the average velocity in the pipe is equal to $v_m/2$. With $v = 0$ at $r = d/2$, (the wall), Eq. 6 by direct substitution becomes

$$0 = 2V - \frac{\Delta p}{4\mu L}\frac{d^2}{4} \tag{7}$$

and the pressure drop per unit length is

$$\frac{\Delta p}{L} = \frac{32\,\mu V}{d^2} \tag{8}$$

which is the familiar expression for laminar flow pressure drop in a pipe.

The relationship expressed in Eq. 8 was used as the criterion in this investigation in order to establish how closely the flow might correspond to that of a Newtonian fluid, even though, by other definitions, the fluid might be considered non-Newtonian.

APPARATUS. The flow system used in the investigation is shown schematically in Fig. 1. The test section consisted of approximately 40 feet of one inch O.D. by .049 inch wall copper tubing to which pressure taps had been attached 35 feet apart. This test section was of a necessity formed to fit into the limited space of the cold box and, therefore, incorporated 5 return bends of 9½ inch radius. A bend radius to pipe diameter of such large magnitude had little significant influence (less than one per cent) on the resultant pressure drop compared to a straight line of the same tubing.[11] Lubricant was pumped around the closed circuit by a gear pump rated at 300 psi and with a nominal flow rate at 3000 rpm of 3.5 gallons per minute. A variable speed drive allowed operation of the pump over a range of 500 to 3600 rpm, thus providing a 7.2 to 1 ratio of flow rates. The inlet to the pump incorporated a section of two-inch diameter glass. The size served the purpose of reducing inlet pressure drop to a minimum, and the transparency allowed visual observation of conditions at entrance to the pump.

The pump system was enclosed by a cold box designed specifically for the purpose and is shown schematically in Fig. 2. Design of the cold box was based on a need for an inexpensive method for providing temperatures down to less than —80 F. In order to obtain such temperatures economically, it was decided to use dry ice as a cooling medium, placing about 400 pounds on one side of a barrier dividing the box and circulating the resulting low-temperature atmosphere around to the other side of the barrier. A low-speed "transfer" blower was used for this purpose. The cold box thus was in essence a low-velocity "cold tunnel." Temperature control was effected by thermostatically controlled operation of the transfer blower.

The material under test was stored in a supply tank at a slight elevation to the pump. At the end of the system the oil was collected in a measuring tank which was slightly elevated to the supply tank. The system held about fifteen gallons. Flow

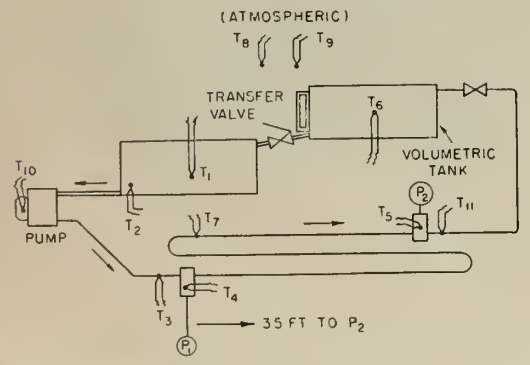

Fig. 1. Schematic layout of system.

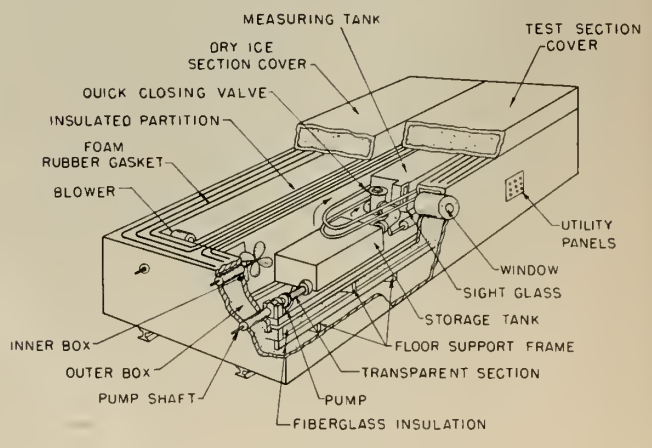

Fig. 2. Cold Box and apparatus.

Table I. Properties data as measured.

Table I Lubricant	Temp. °F.	Viscosity Cs.	Temp. °F.	Specific Gravity	Pour Point °F.
MIL-O-6081B 1005 grade	100	5.23	168	0.822	
	68	8.63	118	0.841	
	-65	2679	80	0.852	
			33	0.871	below -80
			15	0.878	
			-4	0.885	
MIL-0-6081B 1010 grade	100	10.18	60	0.873	
	68	22.4	25	0.886	below -70
	50	36.3	0	0.895	
	-40	2810	-25	0.903	
"Hi-pour" 1005 grade	130	4.41	170	0.822	-15
	100	5.29	90	0.850	
	-10	83.9	61	0.860	
			33	0.871	
SAE 10W	210	6.0	192	0.831	
	130	19.6	137	0.850	
			110	0.858	-30
	100	38.0	80	0.870	
			45	0.882	
Di(2-ethyl-hexyl) sebacate	210	3.3	171	0.873	
	100	13.1	124	0.892	below -70
	-40	1,740	71	0.911	
	-65	10,740	27	0.928	

so-called viscosity pour point, that for the "hi-pour 1005" was a gel and was due to a crystalline structure, composed of a waxy portion of its composition, restraining gravity movement of its remaining components. Once the gel structure was formed and broken, the crystals of which it was composed had little detectable influence on the flow characteristics.

CONCLUSIONS. 1. Pumping of a gas turbine lubricant at temperatures near the pour point presents no difficulty as long as the inlet system is sufficiently generously designed to preclude cavitation at that point.

2. The low-temperature flow characteristics of fluids representative of aircraft turbine applications are as predicted according to laminar flow or Newtonian flow calculations.

3. It is possible under certain conditions to circulate some lubricants at temperatures well below their so-called pour point. These are materials having gel, as contrasted to viscosity, pour points.

4. The flow characteristics below the gel pour point of a representative fluid corresponded to what could be expected from an extrapolation of higher-temperature viscosity.

5. A representative fluid with a viscosity pour point could not be satisfactorily pumped at temperatures within 10 degrees of its pour point. Flow at any condition was according to laminar predictions.

6. Pour point is a poor criterion for predicting the lower limit of pumpability. It has been demonstrated that gel structures which define a pour point have little influence in relation to forced flow. Conversely, viscosity pour points definitely limit the pumpability range.

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drop is also not highly practical. The data on grade 1010 oil were the first obtained in the investigation. Later tests on the other lubricants incorporated an improved technique, with the result that the divergences in the low-temperature region were not so evident.

Each of the lubricants tested was treated in a manner similar to that described for the grade 1010 oil. For an overall comparison, the results were expressed by a transposition of Eq. 8 to the form

$$\frac{\Delta p}{LV} = \frac{32\,\mu}{d^2} \qquad (9)$$

Since the diameter of the line was a constant, 0.902 inches, the results for each lubricant can thus be expressed as a single curve. These five computed curves are shown in Fig. 6 with the corresponding experimental data. Fig. 6 can then be used for a gross assessment of the phenomenon under study.

Examining the performance of the SAE 10W oil, which had a pour point of —30 F., shows that the flow is, as expected, according to laminar flow relations down to approximately 10 degrees above the pour point. No data could be obtained closer than 10 degrees above the so-called pour point.

The data for the 1010 and 1005 grade oils shown as open points on Fig. 6 gave similar results to the SAE 10W. The 1010 grade was mobile down to about —65 F., and the 1005 grade to —75 F. Even though the cold box was capable of about —90 F., the temperature of —75 F. was the limit for the oil when heated due to pumping. It is evident that both of these hydrocarbon materials acted as expected according to Newtonian flow predictions down to the lowest temperature limitations of the system and apparatus.

Di(2-ethylhexyl) sebacate yielded results as expected over most of the low temperature range. For some reason as yet not clear, the data for the sebacate diverges at —60 degrees. The expected pressure differentials were not obtained, and the magnitude of the discrepancy is believed to be beyond the expected experimental errors. No obvious physical changes were noted through the transparent sections of the flow system. More effort will have to be expended in further investigation of this apparent flow anomaly.

The most interesting behavior was that of the "hi-pour 1005" oil. As is apparent from the closed points on Fig. 6, this fluid was capable of being pumped down to —75 F., well below its —15 F. (nominally —20 F.) pour point. For purposes of comparison, the predicted flow characteristics in the region below the pour points were based on an extrapolation of the higher-temperature (33 F. to 170 F.) viscosity measurements. Obviously, viscosity measurements taken by gravity potentials at the lower temperature would have no meaning, even if obtainable. The data for the "hi-pour 1005" grade oil fall slightly above the predicted curve in the low-temperature range. The significance of this divergence is questionable, since the extrapolated "viscosity" could well have been largely influenced by relatively small errors in the determination of the high temperature viscosity. There is also question that the viscosity-temperature curves in the low-temperature region correspond to straight lines obtained through extrapolation, especially near the pour point.

The behavior of the "hi-pour 1005" oil, while not entirely as expected, is a result of the character of its pour point. Whereas the SAE 10 oil had a

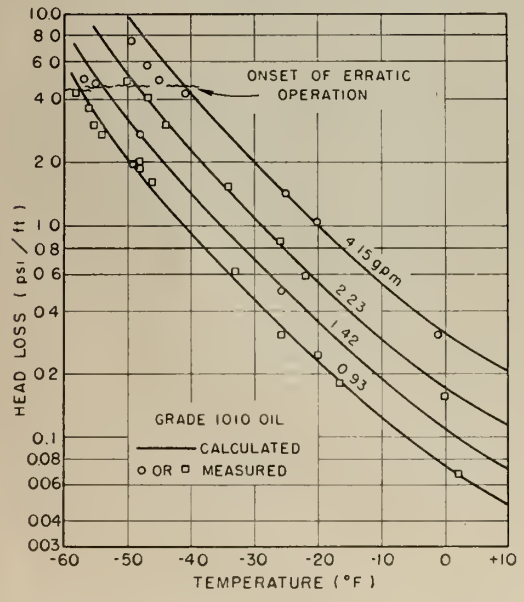

Fig. 5. Flow characteristics for grade 1010 lubricant.

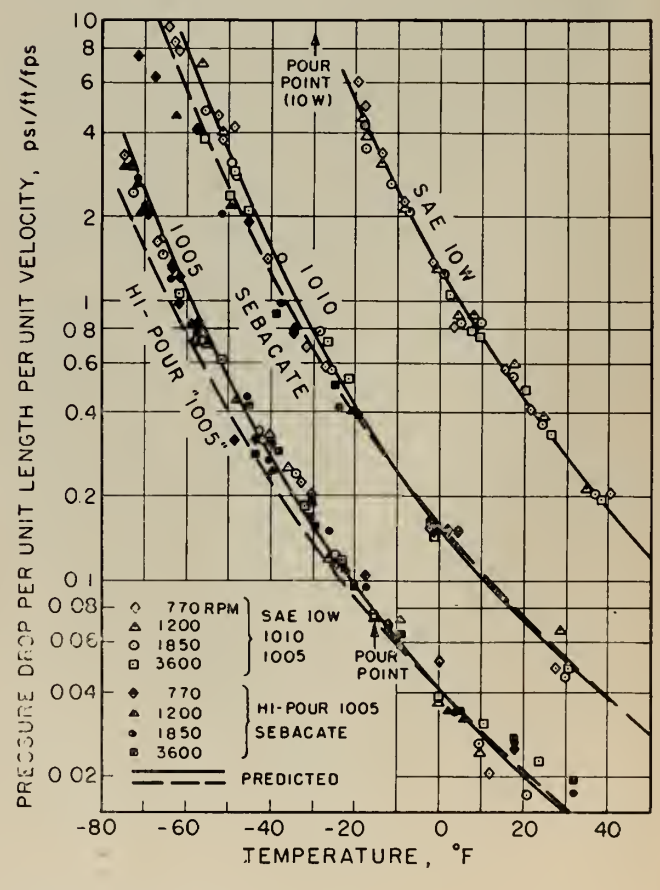

Fig. 6. Combined results.

rate was computed from the measured time for oil to accumulate between calibrated levels in the volumetric tank. During this period the air-operated valve was closed between the two tanks. At the end of a run, the dump valve was opened to allow flow into the supply tank by gravity. During any test the pump was run continuously. The influence of shut-down or soak period was not determined as a part of this study.

The biggest problem to be faced in an investigation of low-temperature pumpability arises from the low mobility of the fluid in the pressure tap lines. Under some conditions these lines may even be at a lower temperature than the portions of the system under test. The speed of response of the pressure-sensing instruments is, therefore, usually unsatisfactory if not altogether impossible.

For this investigation pressure taps and lines were devised as shown in Fig. 3. These pressure taps were made from bakelite and clamped on to the tubing with O-ring seals. Quarter-inch copper lines were inserted into the bakelite to a point just short of the test tubing. These copper pressure lines were then heated by resistance wire to provide fluidity of the lubricant which they contained. However, only enough heat was wanted to allow satisfactory measurements. In order to insure that this heating of the pressure tap lines did not influence the flow characteristics, thermocouples were inserted into the taps at a point near the test section wall, and the current flow to the resistance wire controlled so that the temperature at this point never exceeded the temperature in the flow stream by more than a few degrees. The use of these taps resulted in successful and dependable pressure measurements over the entire range of temperatures encountered.

Thermocouple temperature measurements were taken at the points shown in Fig. 1. The temperature used for relating flow properties of the lubricant is indicated as T_7.

CHOICE OF LUBRICANTS. The selection of test lubricants was based on a compromise in an effort to acquire the maximum amount of information while restricting the number of fluids to a minimum. Five oils were selected which included two representative hydrocarbons and a synthetic, all of military specification, a specially prepared, so-called "hi-pour" 1005 grade (made by a waxy component and a normal component of proper viscosity), and for comparison purposes an SAE 10W motor oil. The first three had pour points of —70 F. or less. The "hi-pour" lube had a pour point of —15 F., and the SAE 10W had a pour point of —30 F.

These five could be listed as: (1) Turbine Lubricant Grade 1010,[12] (2) Turbine Lubricant Grade 1005,[12] (3) Di(2-ethylhexyl) sebacate,[13] (4) SAE 10W motor lubricant, and (5) A specially prepared oil corresponding to 1005 grade in all respects except for a pour point of —15 F.

Pertinent physical properties of the above will be found in Table I, and the viscosities plotted as Fig. 4.

DISCUSSION OF RESULTS. Typical results for one of the lubricants tested, grade 1010, are shown in Fig. 5. Each of the individual flow rates corresponds to a particular pump speed. The gear pump used acted almost as a positive displacement pump for the fluids tested. The top curve, 4.15 gallons per minute, corresponds to 3600 rpm, 2.23 gallons per minute to 1850 rpm, 1.42 gallons per minute to 1200 rpm, and 0.93 gallons per minute to 770 rpm. The solid lines in Fig. 5 were calculated from Eq. 8 and the properties of Table I and Fig. 4. The correspondence of calculated to measured results serves to indicate that at least for this oil the flow was laminar within the limits of measurement.

The region marked as the onset of erratic operation on each of the constant flow rate curves of Fig. 5 corresponds to a combination of difficulties in accurately determining volumetric flow rate, as well as a tendency for the inlet conditions to the pump to become critical. Those points above 5 pounds per square inch pressure drop per foot of line cannot, therefore, be considered dependable. Fortunately, this order of magnitude of pressure

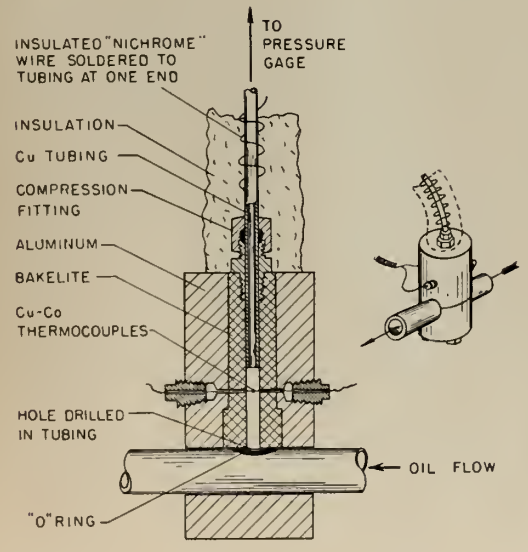

Fig. 3. Heated pressure tap.

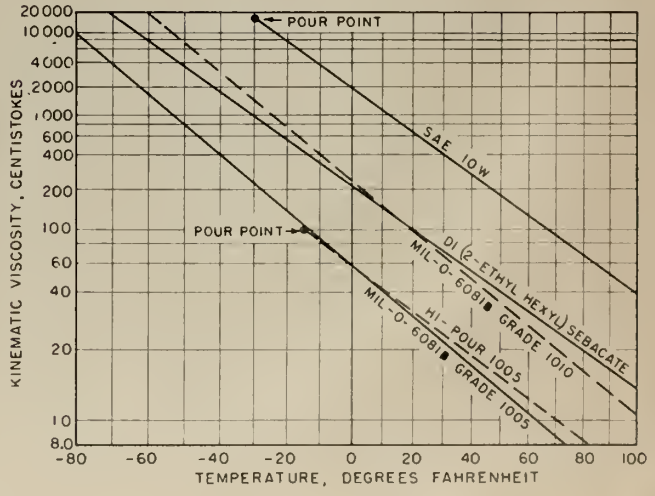

Fig. 4. Lubricant viscosities.

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